Laid-Open Publication

DE 29 26 327 A1

Mechanical/Hydraulic Valve Control

CLAIMS

1. A mechanical/hydraulic valve control for reciprocating piston internal combustion engines with a first piston element allocated to each valve which acts on the valve subjected to force by a closure spring, with a second piston element which can be operated by a cam of the cam shaft, and with a working space disposed between the two piston elements which has a hydraulic pressurizing medium transferring the operating movement of the cam, characterized in that the working space (5) is connected by a control device (6) to a pressurizing medium space (7, 21) which is designed to maintain a constant pressurizing medium pressure.

2. The valve control according to Claim 1, characterized in that the control device is formed by a rotary slide valve (6) which is held on a shaft (33) driven by the number of revolutions of the cam shaft (30).

3. The valve control according to Claim 2, characterized in that the position of the rotary slide valve shaft (33) relative to the cam shaft (30) is changeable dependent upon the operational state of the internal combustion engine.

4. The valve control according to Claim 3, characterized in that in order to drive the rotary slide valve shaft (33), a belt drive (31, 32, 34) driven by the cam shaft (30) is provided onto which an eccentric (40), adjustable dependent upon the operational state of the internal combustion engine, and a tension pulley (35-38) holding the belt tension constant engage.

5. The valve control according to any of Claims 1 to 4, characterized in that the rotary slide valves allocated to the inlet and outlet valves are disposed on a single rotary slide valve shaft (33).

6. The valve control according to Claim 5, characterized in that a slide ring (42), disposed concentrically to the rotary slide valve (6), directly surrounding

the same, is allocated to each of the inlet or outlet valves, which can be turned in order to make relative changes to the valve control times of the inlet and outlet valves with respect to one another dependent upon the operational state of the internal combustion engine.

7. The valve control according to any of Claims 1 to 6, characterized in that the pressurizing medium space (21, 7) has a pressure relief valve (14) and is connected by a throttle (10) to the main lubricant line (13) of the internal combustion engine.

8. The valve control according to Claim 7, characterized in that at least one equalization piston (15) adjustable with respect to a return spring (17) in a cylinder (16) is joined to the pressurizing medium space (21).

9. The valve control according to any of Claims 1 to 8, characterized in that the second piston element (4) has a reduced diameter cylindrical extension (11) on its face surface facing away from the valve (2) which moves (or plunges) into a cylindrical housing collar (12) when the valve is closed.

10. The valve control according to any of Claims 1 to 5, characterized in that a pressure relief valve (45), which opens when a predetermined pressure is exceeded, is joined to the working space (5).

11. The valve control according to Claim 9, characterized in that a damping space (54) enclosed by the cylindrical extension (11) of the second piston element (4) and the cylindrical housing collar (12) can be connected to the working space (5) during the opening stroke of the valve (2).

12. The valve control according to Claim 11, characterized in that a connection line (55) leading to the damping space (54) is joined to the rotary slide valve (6) in such a way that its connection to the working space (5) is opened during the opening stroke of the valve (2) and closed during the closure stroke.

13. The valve control according to any of Claims 1 to 6 and 8, characterized in that the equalization piston (15') connects the pressurizing medium space (58) in a first position to the pressurizing medium line (7) and in a second position to a drain line (60).

14. The valve control according to Claim 13, characterized in that the equalization piston (15') has an annular groove (56) by means of which the pressurizing medium line (7) in the first position of the equalization piston can be connected to a line (57) leading to the pressurizing medium space (58).

15. The valve control according to Claim 13, characterized in that the equalization piston (15') has a face side control edge (59) connecting the pressurizing medium space (58) in the second position to the drain line (60).

Mechanical/Hydraulic Valve Control

The invention relates to a mechanical/hydraulic valve control for reciprocating piston internal combustion engines of the kind specified in the preamble of the claim.

The control and operation of the inlet and outlet valves of reciprocating piston internal combustion engines is achieved nowadays almost exclusively by means of mechanical cam drives. This type of valve drive control does not, however, allow any substantial change to the valve control times relative to the piston position during operation. This is particularly disadvantageous if, for example, in order to reduce the fuel consumption, the throttling losses are to be reduced, the process of the so-called elongated extension is to be used, or the valve overlap is to be reduced with lower numbers of revolutions, close to idling, or also for better starting of the engine. In all of these cases, a valve drive with the greatest possible changeability of the valve control times is required.

Combined mechanical/hydraulic valve controls are already known of the design specified in the preamble of Claim 1 (DE-OS 24 48 311). With these, however, the connection of the working space located between the two piston elements to a pressurizing medium storage container is controlled by the position of control edges provided on the piston elements with respect to movable control bushings. These controls however require that the control edges be produced to a high degree of accuracy, and this involves considerable expense.

The object which forms the basis of this invention therefore consists of providing a mechanical/hydraulic valve control with which, by relatively simple and inexpensive means, extensive changeability of the valve control times for optimal adaptation to different operational states of the internal combustion engine is possible.

Fulfilment of this object is achieved according to the invention in that the working space can be connected by a control device to a pressurizing medium

space which is designed to maintain a constant pressurizing medium pressure. Further advantageous embodiments are given according to the sub-claims. The valve control according to the invention makes it possible to extensively adapt the valve control times to the requirements of the internal combustion engine given by the respective operational states. This is achieved by changing the relative angle position of the rotary slide valve shaft to the cam shaft, it being possible to change the time of opening or of closing of the valves as required.

In the drawings, examples of embodiments of the invention are shown which are described in greater detail in the following.

Figures 1-4	show the schematic structure of the valve control according to the invention with different positions of the cam shaft,
Figure 5	shows the piston movements of the valve control in a diagram with which on the ordinate the piston path and on the abscissa the time in degrees of crank angle are plotted,
Figure 6	shows a schematic illustration of the drive of a rotary slide valve shaft according to the invention,
Figure 7	shows another embodiment according to the invention with which a slide ring is provided in addition to the rotary slide valve,
Figure 8	shows an embodiment of the valve drive with a further peculiarity,
Figure 9	shows a diagram of the cam and valve drive of the embodiment according to Figure 8, and
Figure 10	shows a further variation of an embodiment of the valve drive.

The valve control shown in Figures 1 to 4 for the valves of a reciprocating piston internal combustion engine essentially has the same elements, which are simply in different positions. The numeral 1 here indicates a cam provided for operating a valve 2, which cam does not, however, act directly upon the valve 2, but first of all upon a first, flat-base tappet-type piston element 3 which is held movably in a cylindrical borehole of a cylinder head 22 enclosing a working space 5. A second flat-base tappet-type piston element 4 is connected to the

shaft of the valve 2 and defines the working space 5 downwardly. This second piston element 4 is subjected to force by a valve closure spring 8 in the closure direction of the valve 2, whereas a second pressure spring 9, which is weaker with respect to the valve closure spring 8, is provided between the two piston elements 3 and 4, which pushes the first piston element 3 against the course of the cam 1.

The working space 5 is connected by connecting lines 18 and 12 and a control device in the form of a rotary slide valve 6 disposed between these connecting lines to a pressurizing medium line 7 in which an approximately constant pressurizing medium pressure is permanently maintained. This pressurizing medium line 7 is connected by a throttle 10 to the main lubricant system 13 of the internal combustion engine, and has a pressure limitation valve 14 which is activated when a predetermined pressure is exceeded. Furthermore, the pressurizing medium line 7 is joined to one or more cylindrical spaces 16 in each of which an equalization piston 15, subjected to force by a spring 17, is movably held.

Dependent upon the position of the rotary slide valve 6 and the control edges 19 and 20 attached to the same, the connecting line 18 is connected to the connecting line 21 and so to the pressurizing medium line 7, or separated from the same. The rotary slide valve thus determines the transfer of the operating forces from the cam 1 to the valve 2, and so the control times of the same, because with the closed position of the rotary slide valve 6 the cam movement 1 via the first piston element 3 and the incompressible, hydraulic pressurizing medium of the working space 5 is transferred directly to the piston element 4 and to the valve 2 connected to the same, whereas with the open position of the rotary slide valve the valve 2 is moved by the effect of the valve closure spring 8 towards the closure, the quantity of pressurizing medium pushed out of the working space 5 bringing about an adjustment of the equalization piston 15 by a corresponding amount.

The illustration in Figure 6 shows in which way the rotary slide valve shaft 33 having the rotary slide valve 6 is driven by the cam shaft 30. A cogged belt drive is used here which consists of a drive wheel 31 attached onto the cam shaft 30, a cogged belt 32 and a driven wheel 34 which is attached onto the rotary slide valve shaft 33. The tension of the cogged belt drive is brought about by a tension pulley 35 which is held on a support arm 36 guided in a bearing support 38, which is under the influence of a pressure spring 37 acting upon the pulley 35 in the direction of the tension. On the side of the belt drive lying opposite the tension pulley 35, an eccentric 40 is held on an axis 41. By adjusting the eccentric 40, the relative angle position of the rotary slide valve shaft 33 can be changed in relation to the cam shaft 30. In this way, commencing at the position shown in the drawing, the rotary slide valve shaft 33 is adjusted with respect to the cam shaft 30 in an anticlockwise direction by turning the eccentric in one or the other direction, the tension pulley 35 equalising the length of belt released by the eccentric 40 so that the course indicated by the dot and dash lines is produced in relation to the belt position indicated with full lines.

The position of the rotary slide valve can therefore be changed in relation to the position of the cam in the way described above, for example dependent upon the operational state of the internal combustion engine, such that the control times of the valves can be changed during operation. It is advantageous here if the rotary slide valves allocated to all of the valves are disposed on a single rotary slide valve shaft, as presumed by the illustration in Figure 6. Just one single rotary slide valve shaft 33 is namely provided on the engine housing 39 here. This embodiment would however result in the control times of the inlet and outlet valves being changed in the same way, and this is sometimes not necessarily wise or advantageous. In order to achieve a different changeability of the valve control time here, two separate rotary slide valve shafts could be provided, one of which drives the rotary slide valves of the inlet valves, and the other drives the rotary slide valves of the outlet valves. Another simple possibility for designing the control times of the two valve types differently consists of disposing a slide ring 42 concentrically around the rotary slide valve

6' of the one valve type according to the illustration in Figure 7, said slide ring having a peripheral opening 43 corresponding to the line connector 18' and a second peripheral opening 44 corresponding to the line connector 21'. This slide ring 42 could be turned here by a specific amount, by means of which the opening times of the rotary slide valve 6' are changeable in the sense that with a turn of the slide ring in the clockwise direction, the opening happens later, and with a turn in the anti-clockwise direction, the opening happens earlier.

The function and mode of operation of the arrangement according to the invention will be described in the following using the different positions of the valve drive parts shown in Figures 1 to 4, and using the diagram shown in Figure 5. Commencing with the position shown in Figure 1, with which the valve 2 is in its closed position in which it just still blocks the intake and outlet channel 23 opening out into the combustion chamber 24, by means of a further turn of the cam shaft by fully raising the cam 1 on the first flat-base tappet-type piston element 3, it is shifted towards the valve 2. Because in this position the working space 5 is blocked by the rotary slide valve 6 which is still in a closed position, this movement of the piston element 3 is transferred, unchanged, to the piston element 4, and so to the valve 2, so that the valve is opened.

In Figure 2 the position is shown in which the rotary slide valve 6 is just still closed, whereas several angle degrees later, the control edge 19 opens the connection of the line connector 18 to the line connector 21, and so to the pressurizing medium line 7. The effect of this opening is that a part of the pressurizing medium volume displaced by the first piston element 3 which becomes larger and larger as the opening increases, is pushed into the lines 21 and 7 and pushed into the cylinder as the equalising piston 15 is displaced, and so is no longer used in order to adjust the second piston element 4 and the valve 2 towards the opening.

Finally a point is reached at which more pressurizing medium from the working space 5 is pushed via the open rotary slide valve 6 into the pressurizing medium system 7 which substantially remains at constant pressure than

corresponds to the displacement of the first piston element 3 by the cam 1. At this moment, the valve 2 begins to close under the force applied by the valve closure spring 8 acting upon the second piston element. This closing process starts before the largest lift of the cam has acted upon the first piston element 3, and this position has been reached in Figure 3. Here, as can be seen from the drawing, the valve 2 is also already adjusted by a specific amount in the closure direction.

When the cam shaft and the rotary slide valve shaft are turned further, there is finally an acceleration of the closure movement of the valve 2 due to the return movement of the first piston element 3. This is finally closed before the cam 1 rolls on the first piston element 3 with its base circle, and this has therefore returned to its initial position. During the last part of the return movement of the first piston element 3, the pressurizing medium flows back from the line 7 and from the cylinder 16 acting as a store through the connecting lines 21 and 18 back into the working space 5 in order to fill its volume which is increasing again with pressurizing medium. During this pressurizing medium displacement, the rotary slide valve 6 is of course in its open position.

Figure 5 shows a diagram in which the course of the movements of the piston elements 3 and 4 is plotted over time and according to the crank angle. On the abscissa, only the top and bottom dead centers of the working piston of the internal combustion engine in the corresponding cylinder are marked here. From this it can be seen that the diagram is drawn for the example of an inlet valve which opens approximately at the top dead center (o.T.) and closes at the bottom dead center (u.T.). The curve 25 shows the course of the path of the first piston element 3 operated by the cam 1, whereas 26 shows the piston path of the second piston element 4 connected to the valve 2. With the curve 27 plotted under the abscissa, the position of the rotary slide valve 6 is also indicated, this having its open state in region 27a and its closed state in region 27b, the degree of the opening being of different size due to the constant turning of the rotary slide valve.

It is clear from this diagram that the paths of the first and second piston element run together until the rotary slide valve 6 opens, and that in the further course, the piston element 4 quickly reaches its maximum piston stroke, in order to then fall quickly back to zero, a long time before the piston element 3 also comes back to its initial position.

26' indicates yet another piston stroke course with dot and dash lines, and for the case where by changing the angle position of the rotary slide valve 6 in relation to the cam 1, the start of the latter's opening is shifted in the "later" direction. Because of this there is a greater amplitude of the piston path of the second piston element; moreover the second piston element also reaches its initial position corresponding to the closure position of the valve later so that the outlet valve remains open longer in this case, i.e. for example beyond the bottom dead center, and this can be advantageous, for example in terms of performance gains, with certain operational conditions of the internal combustion engine by means of valve overlaps.

By corresponding control of the rotary slide valve, the control times of the valves can thus be adapted to the respective operational conditions of the internal combustion engine better than previously. Thus, for example, by an appropriate design of the valve drive according to the invention, i.e. of the cams and the rotary slide valves, provision can be made such that with lower numbers of revolutions and with partial loads, no or only very slight valve overlaps, and on the other hand with greater loads and numbers of revolutions, greater valve overlaps are provided. In this way, high torques can be realised both with higher and also with lower numbers of revolutions. Other design criteria could be the best possible partial load consumption, the most stable idling possible with the smallest possible fuel consumption, and good start-up of the internal combustion engine, even with low external temperatures. In addition, the rotary slide valve shaft must be adjusted with respect to the cam shaft, dependent for example upon the number of revolutions or the performance of the internal combustion engine, but other control values are also conceivable.

Finally, a further improvement for still more extensive adaptation of the valve control times to the conditions and to the requirement of the internal combustion engine could be achieved by separate control of the inlet and the outlet valve drives, by providing a rotary slide valve shaft both for the inlet and the outlet valves, or also for example by fitting special slide rings concentrically to the rotary slide valves of the inlet or the outlet valves, the control times of these valves being changed largely independently of one another, as already indicated above.

In order to limit the maximum valve stroke, without at the same time also limiting the valve opening times, according to Figure 8 an embodiment could also be provided with which a pressure relief valve 45 is joined to the working space 5 via a line 46. The valve body 46 of this pressure relief valve subjected to force by a spring 47 opens the access to a bypass 49 via which excessive pressurizing medium flows off into the return when the lower piston element 4 strikes its base as a result of a rise in pressure. In this way, with a maximum valve stroke, which remains constant, practically all of the cam operation time is used up, only the valve opening time, but essentially no longer the valve stroke, being determined by earlier or later engagement of the rotary slide valve. A diagram of the valve and cam strokes for this embodiment is shown in Figure 9 in which 50 shows the course of the cam movement, and 51 shows that of the valve movement. The broken line 52 shows the course of the valve movement with earlier opening of the rotary slide valve, whereas the dot and dash curve 53 shows the course of the valve stroke without stroke limitation.

Finally, in Figure 10 two possibilities for change undertaken with respect to the embodiment according to Figure 1 are indicated in one variation of an embodiment. First of all a line 55 is shown which connects a space 54 enclosed by immersing a cylindrical extension 11 of the second piston element 4 into a cylindrical housing collar 12 to the rotary slide valve 6. The line 55 is joined to the rotary slide valve 6 here such that it is closed during the closure movement of the valve 2, yet connected to the line 18 and to the space 5 during the opening movement of the valve 2. The effect of this is that during the

closure of the valve, hydraulic damping is achieved, yet when opening the valve, an excessive drop in pressure in the space 54 and so the creation of vapour locks, which could have a negative effective upon the valve movement, is avoided.

Figure 10 also shows that the pressurizing medium line 7 does not need to be permanently connected to the line 21, but that it can also be advantageous to allow the connection of these two lines and so the pressurizing medium supply from the equalization piston 15, to be controlled. In addition, the equalization piston shown in Figure 10 by 15' has an annular groove 56 by means of which the pressurizing medium line 7 is connected with a corresponding position of the piston 15' to a line 57 leading to the line 21. At the same time a face side control edge 59 of the equalization piston 15' controls the connection of the working space 58 to a drain line 60. The control is implemented here such that the equalization piston oscillates (or reciprocates) around a central position which lies between the positions in which the working space 58 is connected to the pressurizing medium supply line 7 and to the drain line 60. In this way, the throttle 10 and the pressure relief valve (or pressure limitation valve) 14 provided in Figure 1 can be omitted.